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HEAT PUMP FOR THERMAL POWER PRODUCTION IN DAIRY FARM

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ABSTRACT: Besides cold for milk cooling, dairy facilities need to produce hot water for cleaning of tools and equipment. Overall, ohmic heating has been used in dairy farms, increasing power consumption and manufacturing costs. Therefore, as an alternative to reduce power consumption, this paper proposed a water-water heat pumping for simultaneous cold and heat generations. Accordingly, operational tests were performed with three heat-pump prototypes designed for dairy farms, in both laboratory and field levels. At laboratory, tests were carried out using electricity and CNG to define a coefficient of performance (COP). Biogas tests were performed in the field to measure its consumption. CNG average consumption was of 1.118 m^3/ h, while biogas consume was of 2.02 m^3 / h. COP averages of CNG driven pump were 0.20 for cooling, 0.39 for heating, and 0.59 for global. For electric-power driving, COP values were 1.75 for cooling, 2.25 for heating, and 4.00 for global. In addition to evaporating and compensating temperatures, engine rotation was one factor of influence on heat-pump performance.

KEY WORDS: thermal power, thermal-accumulation, cooling, heating.

INTRODUCTION

In Brazilian dairy plants producing pasteurized milk of types A and B (classification as bacterial count), after milk processing, milking, cooling and pasteurization equipment, as well as facilities and tanks undergo cleaning and sanitizing with the use of hot water between 50 to 60 °C.

BALDASSIN JR. (2006) assessed electric power in processing of “A” type milk, in Campinas-SP, Brazil. They observed a daily energy consumption of 1,064 kWh to produce 4,000 milk liters, wherein had the following shares: cooling (23.01%); pasteurization heating (16.1%); water heating for sanitizing and cleaning (6.19%); refrigerated storing (7.02%); other uses as milking, aeration, bulk tank (47.68%). Therefore, among all energy spent, 30.04% was consumed in cooling processes and 22.28% in heating ones.

Modern dairy units have been partly using their thermal generation capacity from cooling systems seizing cold effects. Broadly, electric resistors are used for heating process, which increases energy waste, since electric heaters consume a lot of energy compared to heat pumping (SHARAF-ELDEEN, 2010).

A few studies have already demonstrated the potential of heat pumps for water and space heating with simultaneous cooling and heating (BYRNE et al., 2009; PARDO et al., 2011; SEBARCHIEVICI & SARBU, 2015). MA et al. (2014) proposed to use solar energy heat pump systems in farms, however, exploiting either heating or cooling individually. Moreover, some heat pump systems have been studied for home application; yet few studies have reported results in farming areas by using distinct sources of energy (STAFFORD & LILLEY, 2012; LUO et al, 2015; VIEIRA et al., 2015).

Dairy plants are prospective customers for heat pumps, since they can harness heat from condensation of milk storage/ cooling system, using it to heat water for cleaning and other production process, reducing electric power consumption.

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Recently, alternative energy sources have been investigated for stationary engines to drive heating (FERNANDEZ et al., 2010; HESARAKI et al., 2015) and cooling systems (FERRAREZ et al., 2010). Dairy farms have plenty availability of confined animal waste, which can be used for biogas production, being thus a source of renewable energy to drive heat pumps. CERVI et al. (2010) and MARTINS & OLIVEIRA (2011) proved biogas feasibility in driving stationary engines for power generation, as a way to increase livestock production sustainability and decentralize energy generation.

In this context, this study aimed to evaluate the performance of three heat pump prototypes, designed for dairy units as for milk cooling and hot water production for cleaning. Heat pumps were driven by electricity, compressed natural gas (CNG) and biogas.

MATERIAL AND METHODS

Aiming to reduce the power of cooling compressor, heat pumps were designed to operate through thermal storage in secondary fluid, accumulating thermal load for 10 hours (milking interval), which is enough to cool 600 liters of milk, corresponding to the largest milking of a dairy farm producing 1,000 liters of milk daily. The fluid used in the cooling circuit of the heat pump was R22.

In designing, energy balance equations were used for cooling cycle and thermodynamic tables for R22. Component selection (evaporator, compressor, condenser and expansion valve) was made through technical catalogs of manufacturers.

Three heat pump prototypes, namely B1, B2 and B3, were assembled, using two cold thermal storage settings. B1 operated with an ice bank, coil type evaporator with 45 m bare copper tube of 15.6 mm diameter. Heat pumps with brazed plate evaporator, using a water-ethanol solution (20% ethanol, mass concentration), were subsequently assembled and named B2 and B3. All the prototypes were built with brazed plate condenser, with heat accumulation in hot water.

Table 1 displays characteristics (type, brand, model and rated capacity) of the cooling components selected for each prototype (B1, B2 and B3). Excluding evaporator, the other components of B1 heat pump were the same as those of B2 and B3 prototypes.

TABLE 1. Cooling components selected for B1, B2 and B3 heat pumps.

Prototype B1				
Component	Type	Brand	Model	Rated capacity (kW)
Compressor	Open	Bitzer	Block III	4.63(*)
Expansion valve	Thermostatic	Danfoss	TEX2-1.5	4.80
Condenser	Brazed plates	Apema	WP 4 -30	6.98
Evaporator	Copper coil	-	-	5.23
Combustion engine	Single cylinder	Honda	GXV240	5.89
Electric engine	Three-phase	Weg	-	2.21
Prototypes B2 e B3				
Component	Type	Brand	Model	Rated capacity (kW)
Compressor	Open	Bitzer	Block III	4.63(*)
Expansion valve	Thermostatic	Danfoss	TEX2-1.5	4.80
Condenser	Brazed plates	Apema	WP 4 -30	6.98
Evaporator	Brazed plates	Apema	AE 5-20	5.23
Combustion engine	Single cylinder	Honda	GXV240	5.89
Electric engine	Three-phase	Weg	-	2.21

* Rated cooling capacity under operation at evaporation temperature of -5 ºC and condensation one of 30 ºC.

To drive compressor, considering conversion losses from petrol to gas, an air-cooled stationary engine (GXV240 Honda) was selected, delivering 5.89 kW (8 hp) of power. Figures 1 and 2 show assembling schemes among B1, B2 and B3.

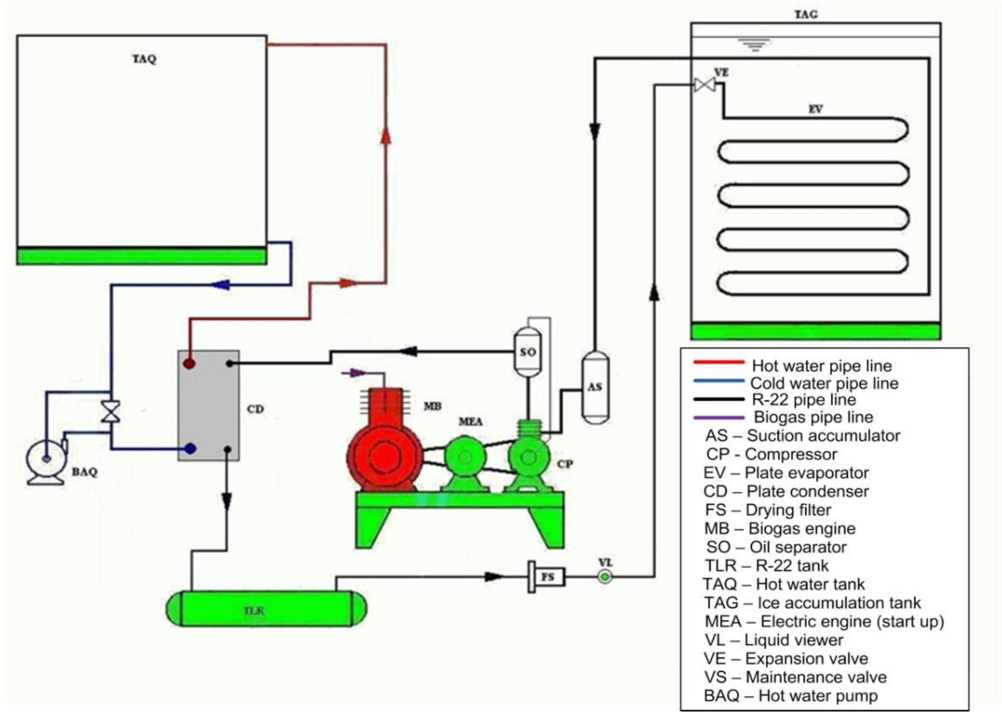

FIGURE 1. Assembling scheme of B1 prototype.

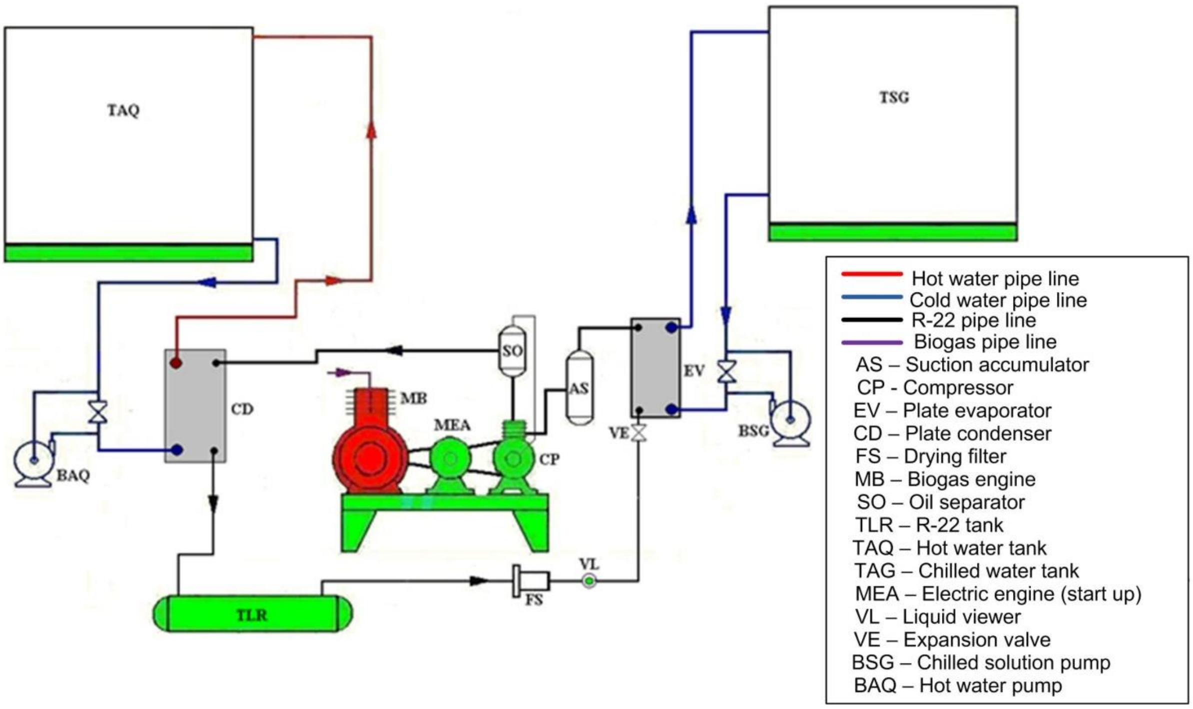

FIGURE 2. Assembling scheme of B2 and B3 prototypes.

Combustion engine was started by using electric motors with power enough for driving compressors, also aiming to use them in electric drive testing of the heat pumps.

B1 was designed to deliver 219 kg ice 415 liters cold water (0 ºC) and 1,000 liters hot water (60 ºC) each operation cycle (10 hours). Yet B2 and B3 were developed to produce the same amount of hot water as B1 does, besides cooling 2,000 liters of water-ethanol solution up to -9 ºC. Figure 3 displays coil layout (evaporator) for B1.

FIGURE 3. Coil system (evaporator) designed for B1 heat pump.

Figures 4 and 5 show how heat pumps were installed; among which B1 and B3 were assembled in laboratory for electricity and compressed natural gas (CNG) tests, while B2 was set in the field for biogas testing.

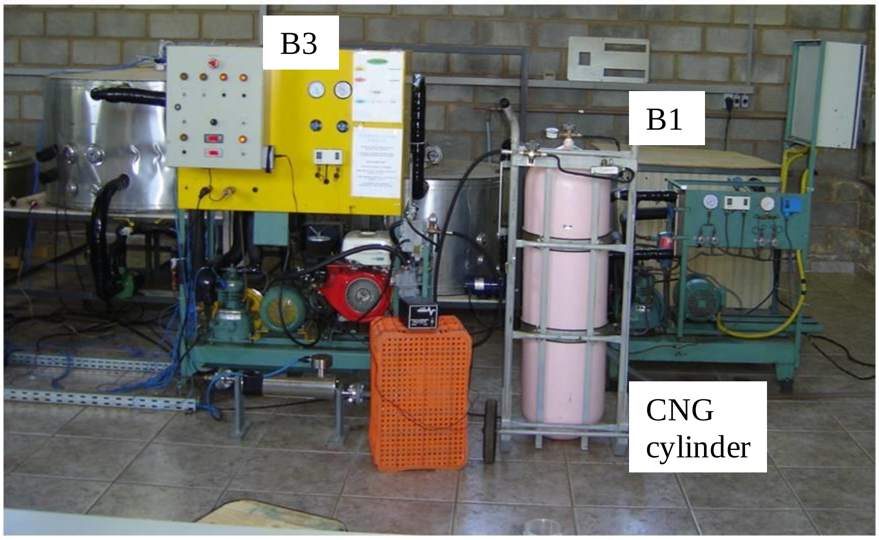

FIGURE 4. Heat-pump prototypes B1 and B3 installed in laboratory for electricity and compressed natural gas (CNG) tests.

FIGURE 5. Heat-pump prototype B2 assembled in the field for biogas testing

Field-testing was performed at an experimental field for Biogas Production belonging to the Faculty of Agricultural and Veterinary Sciences, University of São Paulo State - UNESP, in Jaboticabal city. Laboratory testing was carried out at the Laboratory of Thermodynamics and Energy of the Faculty of Agricultural Engineering, University of Campinas - UNICAMP, in Campinas city. Both in São Paulo state, Brazil.

As B1 and B3 were mounted one next to the other and did not run simultaneously, both pumps shared the same combustion engine. In order to reduce costs, B2 and B3 reservoir volumes were halved, being 1,000 liters for water-ethanol solution and 500 liters for hot water. Consequently, operating times were also halved (5 hours) to achieve a same temperature profile. As result, coefficient of performance (COP) data could be extrapolated as for a full-tank capacity.

For B1, due to the ice bank behavior, reductions on operation time were precluded for further use of the hot-water tank with smaller volume. Thus, we kept a 10-hour operation time to prevent any interference with COP. Therefore, the water-ethanol tank from B3 was drained out during the tests, being used as a hot water reservoir for B1.

For CNG testing, we used a gas conversion kit that comprises a pressure-reducing valve and a 21-m^3 high-pressure cylinder (20 MPa), which was assembled on a cart for easy transport. Since this cylinder owned a supply valve which enabled it to be supplied in gas stations.

Unpurified biogas was used in B2 field tests; this gas comes from gas meter powered by two digesters, an Indian and a Chinese model. In this case, pressure-reducing valve was discarded, since biogas pressure was low.

B1 and B3 were equipped to measure the following parameters: temperature, pressure, refrigerant mass flow rate, water flow in heat exchangers, gas consumption of combustion engine, and electric engine power/ consumption, in the case of electric driving of heat pumps.

COP was evaluated for B1 and B3 individually; this index relates thermal energy generated and consumed. Additionally, such index was also calculated for the engine-compressor set, disregarding consumptions of auxiliary equipment.

In the case of B2, as biogas volume stored in gasholder was of only 4 m^3, tests lasted less than 2 hours, hampering COP estimation. Thus, field tests helped assessing biogas consumption and compare to CNG intake for B3, once this prototype was identical to B2. This condition enabled us inferring that both devices have the same COP; ergo there was just a consumption difference when using a gas with lower calorific value.

Regarding the electric engine driving, instantaneous COP estimates for heating and cooling pumping (COP_{cool} and COP_{heat}) were calculated, respectively, through eqs. (1) and (2).

$$COP_{refri} = \frac{\dot{Q}_{ev}}{\dot{W}_e} \tag{1}$$

$$COP_{aquec} = \frac{\dot{Q}_{cd}}{\dot{W}_e} \tag{2}$$

where,

COP_{cool} - Coefficient of performance for cooling system [dimensionless];

COP_{heat} - Coefficient of performance for heating system [dimensionless];

$\dot{Q}_{ev}$ - Instantaneous rate of heat removed by the evaporator [kW];

$\dot{Q}_{cd}$ - Instantaneous rate of heat transferred by the condenser [kW],

$\dot{W}_e$ - Electric engine instantaneous power [kW].

For B1 and B3 tests, operating with CNG, instantaneous power ($\dot{W}_e$) was replaced by fuel energy flow ($\dot{E}_{fuel}$), which can be estimated by [eq. (3)].

$$\dot{E}_{fuel} = PCI_{gas}.\dot{V}_{fuel} \tag{3}$$

where,

$\dot{E}_{fuel}$ - Fuel energy flow [kW];

PCI_{gas} - Lower calorific value of natural gas [kJ/m^3];

$\dot{V}_{fuel}$ - Fuel flow [m^3/s].

PCI_{gas} and natural gas composition were provided by COMGÁS (São Paulo Gas Utility Company), being based on natural gas from Bolivia – Brazil gas pipeline, since this was the source of the CNG used in this research. Therefore, we considered an average PCI_{gas} of 36.5 MJ m^{-3} (COMGÁS, 2015).

RESULTS AND DISCUSSION

Figures 6 and 7 show power variation required to drive compressor (electric driving) and natural gas consumption for combustion engine for B1 and B3, respectively, over the operation time. It is clear that in spite of a wide variation, the fitted curve shows increasing trend of natural gas uptake for a combustion engine, due to condensing temperature rise. Nevertheless, when comparing to electric engine power curve, there is a less marked increase in gas consumption.

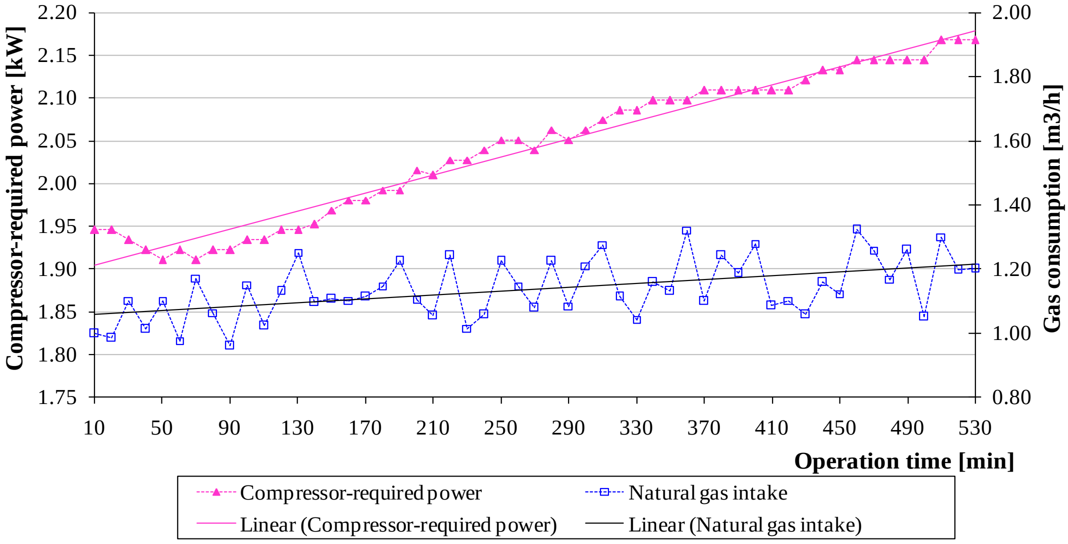

FIGURE 6. Power variation required to drive compressor (electric engine) and combustion-engine natural gas consumption for B1 heat pump over the operation time.

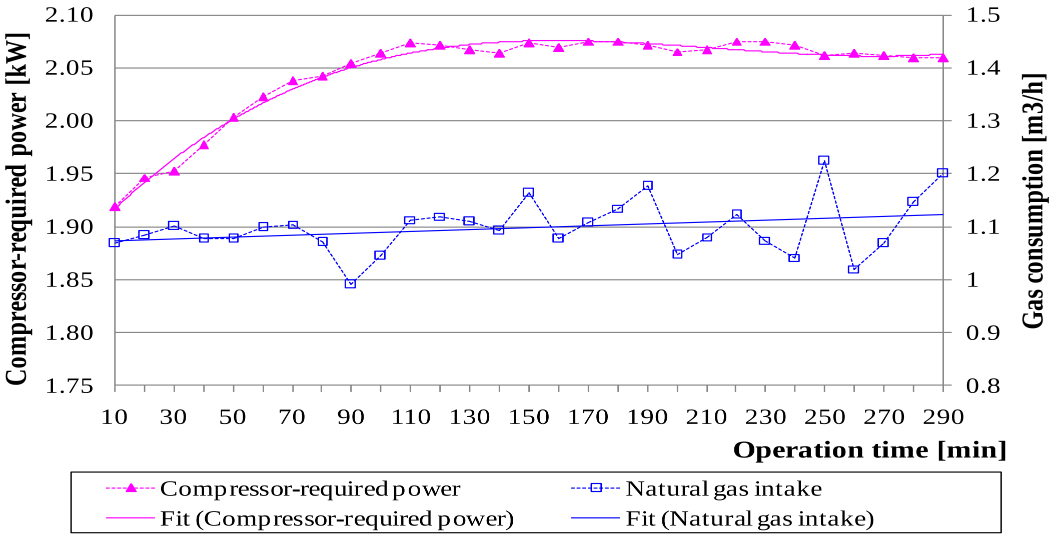

FIGURE 7. Power variation required to drive compressor (electric engine) and combustion-engine natural gas consumption for B3 heat pump over the operation time.

For electric driving tests, the power required by compressor was regarded as the value measured at the engine from the start to the end of the process, this consumption ranged from 15 to 18%. Whereas for the gas engine, under the same conditions, power uptake increase was nearly from 5 to 7%.

In CNG testing, B1 reached an average consumption of 1.145 m^3 h^{-1}, while B3 showed an average of 1.09 m^3 h^{-1}. Testing B2, it was noted an average biogas consumption of 2.02 m^3 h^{-1}, which is 85% higher than CNG one in B3, and 76% greater than in B1.

It is noteworthy reporting that in nature biogas has a lower methane and higher carbon dioxide concentrations if compared to CNG. The first has on average 65% CH_4 and 35% CO_2, showing then an average calorific value of between 20.9 and 29.3 MJ (FERRAREZ et al., 2010). On the other side, CNG is composed of 91.8% CH_4 and 0.08% CO_2. Based on these data, we may state that consumption differences were directly related to their respective calorific value. Consequently, this led to an estimate of 19.11 MJ Nm^{-3} for calorific value of biogas, being close to the lowest value reported by FERRAREZ et al. (2010).

The hourly average biogas consumption of B2 results in a daily intake of 40.4 m^3, within 20 hours of operation, which is an energy demand a dairy farm, producing 1,000 liters of milk daily, would afford. Since to produce a daily average of 20 liters of milk per animal, 50 confined animals would be required. According to NOGUEIRA et al. (2012), this is an amount able of providing a daily production of 50 m^3 of biogas.

In Figures 8 and 9, one can see the curves related to COP of B1 and B3, respectively, as function of the difference between condensing and evaporating temperatures for CNG driven engines. Interestingly, a descending trend is observed due to thermal accumulation, with increasing condensing temperatures and decreasing evaporating ones throughout the process. This outcome highlights an increasing compressor-required power and, consequently, a reduced refrigerating storage capacity.

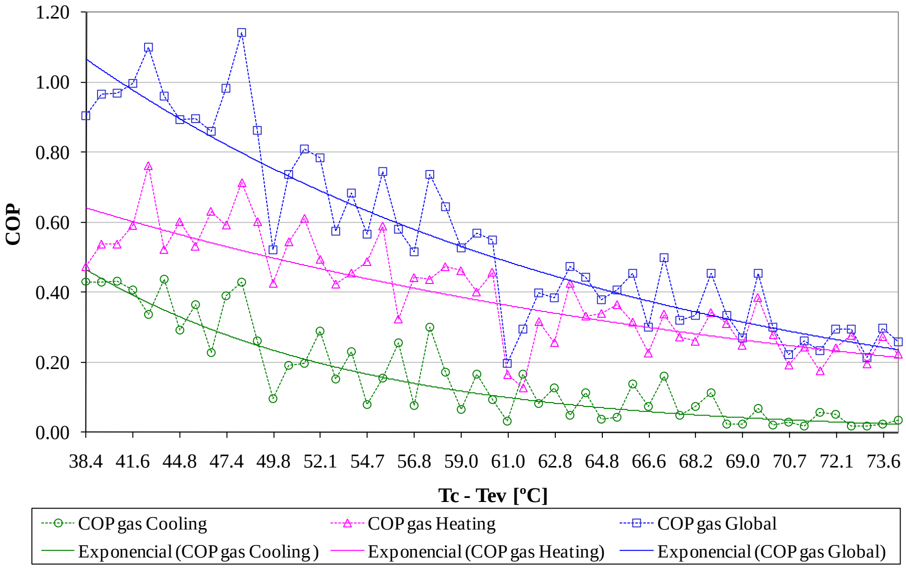

FIGURE 8. Variation on COP values of B1 heat pump as function of the difference between condensing and evaporating temperatures for CNG driven engines.

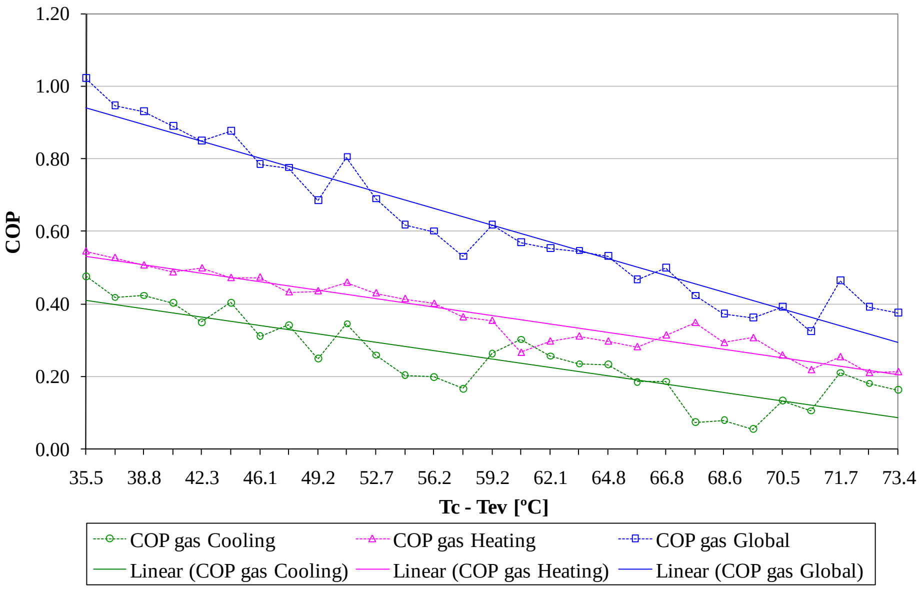

FIGURE 9. Variation on COP values of B3 heat pump as function of the difference between condensing and evaporating temperatures for CNG driven engines.

Table 2 presents the COP averages for B1 and B3, as function of the difference between condensing and evaporating temperatures, for CNG driven engines. Cooling system stood out with the largest differences, wherein B3 surpassed by 60% the values of B1. It might have occurred due to an efficient heat exchange in the plate evaporator, with air-forced circulation, whether compared to the ice bank. The evaporator in B1, as promotes ice accumulation, loses effectiveness throughout the process since ice layer increases, acting as an insulator and reducing heat exchange with the cooling fluid.

TABLE 2. Coefficients of performance for B1 and B3 heat pumps for CNG driven engines.

Pump	Cooling	Heating	Global
B1	0.15	0.37	0.52
B3	0.24	0.40	0.64

Testing the accumulation of gas-fired water heaters adapted to biogas, SILVA et al. (2005) observed an energy efficiency of 68%, which is a performance superior to those obtained by the two heat pumps analyzed here. However, when it comes to cooling and heating equipment and considering the overall COP, which takes into account gas primary energy conversion to cold and heat, the values are quite close to those found by these authors.

Conversely, when dealing with heat generation for refrigeration systems, efficiency tend to be lower. According to PRIDASAWAS & LUNDQVIST (2004), large cooling systems by absorption of high-efficiency, COPs can reach values between 0.6 and 0.8; whereas vapor compression refrigeration systems, fired by steam turbines (Rankine cycle), can achieve values between 0.3 and 0.5.

ROSSA & BAZZO (2009), studying an absorption chiller of 17 kW thermal capacity (three times bigger than B1 and B3) and driven by heat waste from a natural gas turbine, reported COP values between 0.25 and 0.31 for a cooling process. SILVA & MOREIRA (2008) simulated chillers fired by natural gas and landfill gas through EES software (Engineering Equation Solver), and observed COPs from 0.12 to 0.36, for equipment with cooling capacity between 18 and 70 kW. This way, the results obtained here for B1 and B3 in cooling system (from 0.15 to 0.24) seem to be satisfactory.

The low COP values of CNG driven engines occurred, since it was taken the gas as energy source of both prototypes. Thus, this index was affected by a relatively low efficiency of combustion engines. According to the calorific value of natural gas and electric-driven engine tests, which were carried out under the same operating conditions, average engine thermal efficiency was 20%, corroborating results from SOUZA et al. (2010) for small power-generators and single-cylinder engines powered with natural gas and biogas.

To enhance efficiency, higher capacity engines would be necessary, such as those used in cars for example, which can reach higher values being supplied by natural gas. Such engine heat driving heat pumps would be able to increase the COP up to 90%.

Heat recovery might also contribute to increase exploitation of energy provided by burning gas. ELGENDY et al. (2011) and YANG et al. (2013) observed a primary energy ratio (PER) greater than 1.1, using air-water heat pumps driven by water-cooled gas engines, recovering heat lost from the engine block and from flue gases.

Figures 10 and 11 show the curves of COP for electric driving tests for B1 and B3, respectively.

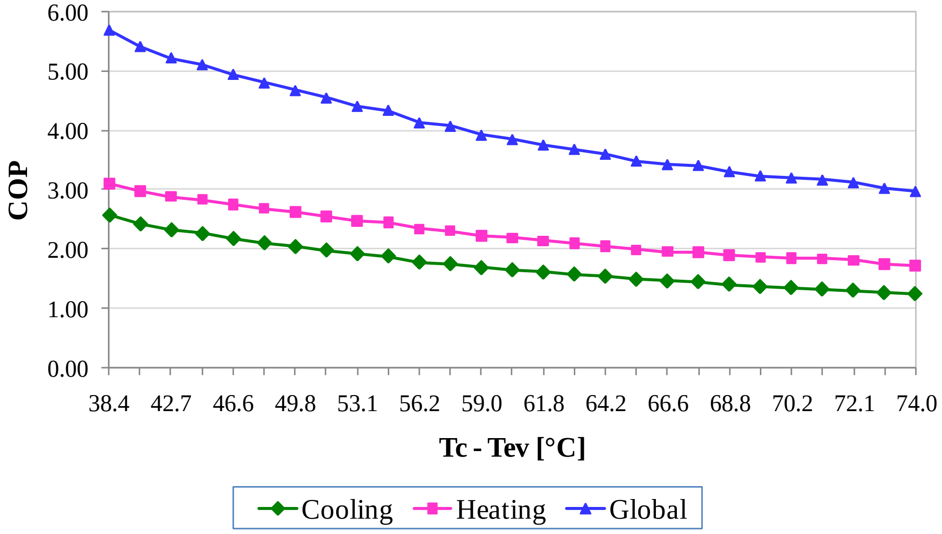

FIGURE 10. Variation of COP in B1 as function of different condensing and evaporating temperatures in electric driving tests.

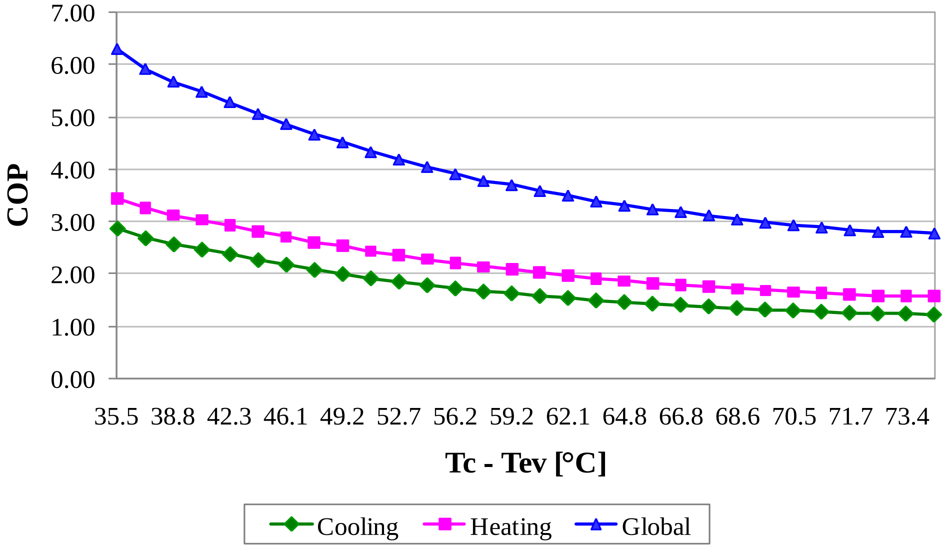

FIGURE 11. Variation of COP in B3 as function of different condensing and evaporating temperatures in electric driving tests.

Table 3 show COP average values for B1 and B3 electric power driven. Comparing to CNG, cooling system COP dropped from 60% to 0.57%, and heating COP was slightly lower than in B3, affecting global value. In this case, electric engine rotation had influence, which justifies the distinct outcomes from natural gas. Given these results, electric motor rotation was measured by a digital tachometer, finding that B3 worked at 1,710 rpm, while B1 rotation was of 1,770 rpm, differing in 3.51%.

TABLE 3. Coefficients of performance for B1 and B3 heat pumps electric power driven.

Heat pump	Cooling	Heating	Global
B1	1.74	2.27	4.02
B3	1.75	2.22	3.97

On one side, aside from the rotation difference, the pulley diameter of electric engine in B1 was larger (155 mm) than in B3 (105 mm), enhancing performance of the first. On the other side, gas engine in B1 had its performance impaired because the electric engine made the driving connection between combustion engine and compressor.

As reviewed by ELGENDY et al. (2011), increasing combustion engine rotation, for heat pump driving, from 1,300 to 1,750 rpm, caused a 17% increase in recovered heat for water warming. In contrast, it also caused a 39% increase in gas consumption, reducing by 15% the COP, called primary energy ratio (PER) by these authors. In this study, B1 and B3 combustion engine rotations were not changed. During the tests, the throttle lever of combustion engine was set at ¾ after its stroke. Rotation variation occurred during changes from gas to electric power.

CHOI et al. (2014) studying heat pumps connected in cascade for warming a greenhouse, supplied by external units removing soil heat to warm water to be used as heat source to the indoor units, which transferred heat to the indoor air, obtained COP values between 2.9 and 3.8. Notwithstanding, these authors have made assessments under less severe operating conditions as those observed here for B1 and B3, with positive evaporation temperatures and condensation temperatures relatively low. This made us conclude that the performance of both pumps with electric driving can be considered satisfactory, if the operating conditions are took into account (low evaporation temperature and high condensation temperature).

CONCLUSIONS

The results showed the technical feasibility of using two thermal effects (cooling and heating) of a heat pump in a single device, exploiting heat normally wasted by conventional cooling systems to heat water used for cleaning dairy facilities. Both heat pump models studied (B1 and B3) reached the thermal power production demand to which they were designed. In the case of hot water generation, were able of providing values above the designed ones.

The coefficients of performance achieved, by both gas and electricity driving, are in accordance with the literature, as long as operational conditions to which the prototypes were submitted are observed. Combustion engine had a relatively low efficiency, what had a significant impact on COP of the pumps.

The performed tests presented the possibility of driving heat pumps with biogas from animal manure, generated within dairy farms. Thus, implying in self-sufficiency in thermal power generation in areas where electricity supply is barely provided. Regarding the engine rotation, the prototype B3 was most effective than B1, therefore, proving that water-ethanol solution promoted greater thermal storage than the ice bank did. Even so, further studies are recommended to better evaluate the influence of rotation on the coefficient of performance (COP) and increasing application range, enabling the use of more efficient engines for higher performance coefficients.

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